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Vibro-acoustic analysis of a coach platform under random excitation

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ABSTRACT

Vibro-acoustic analysis of a rail vehicle cabin is presented in this paper. Vehicle is modeled by an air cavity coupled to a flexible floor panel. Analytical procedure is employed to predict the structural-borne noise in the vehicle model generated by random excitation of the panel. In this study, natural frequencies of the coupled system are obtained and then the sound pressure field inside the cavity is analytically determined. In order to find dynamic responses of the coupled system in the time domain, Durbin's numerical Laplace transform inversion algorithm is employed. Convergence of the algorithm is verified and eventually a parametric study is carried out to investigate the effects of rail irregularity and vehicle speed on the time and frequency responses.

1. Introduction

Vibro-acoustic analysis of plate-cavity systems has been significantly investigated because of variety of engineering applications, in the recent years. Interaction between a flexible plate and a cavity is an attractive problem in many engineering applications. Several cases may be addressed in automotive, railway and aircraft industries. Dynamic coupling between the panels and acoustic enclosures in different vehicles is a recent engineering challenge. Subsequent to this research line, identifying key parameters in the problem of structure-borne noise in rail vehicles is the main objective in the present study. Free and forced vibration analyses of such coupled systems have been addressed in the literature. Pretlove [1] employed an analytical method to study the free vibration of a rectangular panel backed by a closed cavity. Four modes of vibration have been taken into account in this study and it has been shown that a remarkable change can occur for plate vibration due to the presence of an air cavity. Natural frequencies of a plate-cavity system were subsequently obtained by Qaisi [2]. He made an investigation into effects of the cavity depth and boundary conditions on the first natural frequency of such coupled system. Qaisi used a formulation in the matrix form and obtained mass and stiffness matrices for a flexible panel backed by a cavity. Mohamady et al. [3] studied the interaction between a flexible wall and an air cavity by use of the finite element method. Natural frequencies and mode shapes of the coupled system were obtained in that study. Tanaka et al. [4] obtained the natural frequencies of a coupled system by use of different analytical and numerical methods. Sadri and Younesian [5,6] used variational iteration approach to study the effects of different parameters on nonlinear natural frequencies and harmonic responses of a plate-cavity system. Three modes of vibration have been considered in their study and equations of motion have been derived for the coupled system. Investigation of sound absorptions and frequencies of different panels backed by an air cavity has been also addressed in the literature [7–9]. Based on the investigations carried out in the mentioned research studies, for analyzing free vibrations of the plate-cavity systems, there is no discussion on higher modes and frequencies of the system. Also, the effect of presence of a cavity on the natural frequencies of a rectangular plate has been examined in most of these studies, while obtaining the dynamical behavior of cavities influenced by a flexible panel can be of interest for many applications.

Different analytical and numerical techniques have been employed by many researchers to investigate the forced vibration of the coupled systems. Kim and Brennan [10] used the impedance and mobility approaches to predict the response of a structural-acoustic system under a point force excitation. They employed four acoustic and six structural modes for their simulations and compared the analytical results to those obtained by experimental procedure. Effect of an air cavity on the sound field reflected by a flexible panel was studied by Nakanishi et al. [11]. Ding and Chen [12] employed a finite element model to predict the response of a plate-cavity system under a random point excitation. A comparison between sound pressure levels predicted by experimental and

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numerical methods was carried out in that study. It has been shown that the finite element method is more efficient for the low-frequency domain. The structural-acoustic coupling mechanism between a flexible structure and a cavity of semi-infinite size was studied by Kim and Kim [13] using the wave-based approach. A combination of analytical and numerical techniques was employed by Bassyiouni and Balachandran [14] to predict the sound pressure inside a plate-cavity system. Alternate supplementary techniques including Green's function, integro-modal and impedance-mobility approaches have also been addressed in the literature [15–17]. Chen et al. [18] used the finite element method and carried out a sensitivity analysis to determine the sound pressure level inside a structural-acoustic system. Different authors have also studied the forced vibration of circular plate-cavity systems subjected to different types of excitation [19–21]. In most of the previous studies, for investigating the forced vibrations of the plate-cavity systems, harmonic excitations have been employed and response of the models has been obtained for a limited number of frequencies. However, utilizing a random force to excite the system is always interesting for many applications in different industries [22]. Many vibrating modes must be taken into account in this case and convergence of the response should be discussed.

Transmission of noise through a double-leaf skin plate has been analytically investigated by Xin et al. [23]. Sound radiation of a rib-stiffened structure subjected to a harmonic point force has been analyzed by Xin and Lu [24]. It has been found that the coupling between the structural and acoustical systems results in additional peaks and dips in the sound pressure response of the model. Xin and Lu [25] analytically and experimentally examined the sound transmission loss for a system composed of two panels and an air cavity. Also, vibro-acoustic analysis of different double panels has been carried out in some references [26–30].

According to the research studies provided in the literature, it seems that there is no analytical solution to describe the effects of mode number, excitation frequency and vehicle parameters on the structure-borne noise inside a rail vehicle. The aim of the present study is to predict the interior noise generated by a rail vehicle floor under the random excitation of rail irregularities. For this purpose, vehicle is modeled by an acoustic enclosure with a randomly excited flexible floor. According to references [31–33], uncertainty of the track parameters and surface irregularity of the rail are two main sources of random vibration in rail vehicles. Random loads generated by the rail irregularity are applied to floor panel at the location of bogie center pivots. Galerkin–Laplace transform technique is employed as an analytical approach to predict the interior noise. A subsequent parametric study is carried out to examine the effects of different parameters (rail quality, train speed, etc.) on the time and frequency response of the system.

2. Mathematical modeling

2.1. Free vibration analysis

In order to obtain the natural frequencies of the coupled system, free vibration analysis has been carried out in this section. The structural-acoustic system is composed of one flexible plate coupled with a rigid-walls cavity. A schematic of the model is shown in Fig. 1. As can be seen, a vibrating rectangular plate of length, a, and width, b, is coupled with an air cavity of depth, c. Acoustic pressure inside the air cavity is governed by the following wave equation:

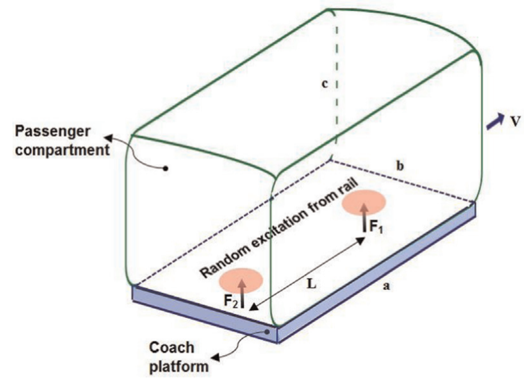

Fig. 1. Schematic of the system subjected to random excitations.

$$\nabla^2 P(x,y,z,t) - \frac{1}{{c_a}^2}\frac{\partial^2 P(x,y,z,t)}{\partial t^2} = 0 \tag{1}$$

where c_a is the speed of sound in the air. Boundary conditions of the air cavity at the flexible and rigid surfaces are given by

$$\frac{\partial P}{\partial x}\Big|_{x=0,a} = 0, \quad \frac{\partial P}{\partial y}\Big|_{y=0,b} = 0,$$
$$\frac{\partial P}{\partial z}\Big|_{z=0} = 0, \quad \frac{\partial P}{\partial z}\Big|_{z=-c} = -\rho_{air}\frac{\partial^2 W(x,y,t)}{\partial t^2} \tag{2}$$

where ρ_{air} is the air density. According to the boundary conditions presented in Eq. (2), the air pressure inside the cavity can be expanded as

$$P(x,y,z,t) = \sum_{r=0}\sum_{s=0}\left(A_{rs}\cos\frac{r\pi x}{a}\times\cos\frac{s\pi y}{b}\times\cosh\lambda_{rs}z\times e^{i\omega t}\right) \tag{3}$$

where $\lambda_{rs} = \sqrt{(\frac{r\pi}{a})^2 + (\frac{s\pi}{b})^2 - (\frac{\omega}{c_a})^2}$ and A_{rs} is unknown coefficient which can be determined using the boundary condition at the flexible wall. The flexible plate displacement at the bottom of the model is governed by

$$D\nabla^4 w(x,y,t) + \rho h\frac{\partial^2 w(x,y,t)}{\partial t^2} = -P_c(x,y,t) \tag{4}$$

where D, ρ and h are the bending rigidity, density and thickness of the flexible plate, respectively. Furthermore, $P_c(x,y,t)$ denotes the acoustic pressure of the air cavity at $z=-c$. It is assumed that the flexible plate is simply supported at its edges and consequently the transverse displacement,$w(x,y,t)$, is given by the following equation:

$$w(x,y,t) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} W_{mn}\sin\frac{n\pi x}{a}\sin\frac{m\pi y}{b}\times e^{i\omega t} \tag{5}$$

We now use the boundary condition at the bottom of the plate-cavity system to solve the problem. Substituting Eqs. (3) and (5) in Eq. (2) results in

$$\frac{\partial P}{\partial z}\Big|_{z=-c} = -\rho_{air}\frac{\partial^2 w(x,y,t)}{\partial t^2}$$
$$\Rightarrow \sum_{r=0}\sum_{s=0}\left[\cos\frac{\pi rx}{a}\cos\frac{\pi sy}{b}\times\lambda_{rs}A_{rs}\sinh\lambda_{rs}c\right]$$
$$= \rho_{air}(-\omega^2)\times\sum_{m=1}\sum_{n=1} W_{mn}\sin\frac{\pi nx}{a}\sin\frac{\pi my}{b} \tag{6}$$

Multiplying the last equation by $\cos\frac{\pi ix}{a}\times\cos\frac{\pi jy}{b}$ and integrating over the plate area, one can derive the following equation: